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"Compressor, especially an axial piston compressor
for a vehicle air-conditioning system"

Description

The invention relates to a compressor, especially an axial piston compressor for a vehicle air-conditioning system, having a housing delimiting a drive mechanism chamber, having a cylinder block in which at least one piston is mounted so as to be axially displaceable back and forth, and having a cylinder head having a suction side and a delivery side.

In specific terms, the compressor is a variable-capacity compressor having a swash plate drive or wobble plate drive which is located within the drive mechanism chamber and by means of which the rotary movement of a drive shaft is converted into an axial reciprocating movement of the piston or pistons. The swash plate or wobble plate, also called in general manner a "tilt plate", is variable in terms of its inclination relative to the drive shaft, which can be coupled to an external motor. The inclination of the "tilt plate" governs the stroke of the piston or pistons. When the pressure in the drive mechanism chamber is relatively low, the inclination of the "tilt plate" is large so that the stroke of the piston or pistons is correspondingly long. When the pressure in the drive mechanism chamber is relatively high, the inclination of the "tilt plate" is small so that the stroke of the piston or pistons is correspondingly short. With respect to the prior art, reference may be made to the following publications:

- DE 196 11 004 A1
- DE 44 41 721 C2
- JP 2002/070739 A

Those publications relate in each case to steplessly regulatable compressors having variable adjustment of the piston stroke.

Such compressors are usually constructed in the form of axial piston compressors, with modification of the stroke - as already mentioned - being accomplished by means of a change in the tilt angle of the "tilt plate". In the process, the position of the lower dead

centre of the piston or pistons is changed; the location of the upper dead centre and, as a result, the size of the so-called clearance volume remains unchanged in the idealised case.

In operation of such a compressor, internal leaks and losses usually occur. The main cause thereof is that, in the course of compression of a coolant drawn into the cylinder or cylinders, a so-called partial mass flow usually enters the drive mechanism space of the compressor through the gap between the cylinder and piston. This effect is also known as "blow-by". Insofar as an oil separator is arranged on the high-pressure side of the compressor, there is a possibility that coolant will, in undesirable manner, enter the drive mechanism chamber by way of the oil return. In order to avoid an undesirable over-pressure in the drive mechanism chamber, there is provided between the drive mechanism chamber and the low-pressure side, or suction side, a fluid connection by way of which leak masses entering the drive mechanism chamber can flow out again. The mentioned fluid connection is usually a connecting bore. The free cross-section of that bore is generally so dimensioned that, even under most unfavourable conditions, no undesirable over-pressure arises in the drive mechanism chamber. Because of the described dependency of the piston stroke on the pressure within the drive mechanism chamber it is usual for the compressor to be externally regulated by influencing the pressure in the drive mechanism chamber. An increase in pressure inside the drive mechanism chamber brings about an effect on the internal force and moment equilibrium of the compressor such that the stroke of the pistons is reduced. The compressor is, as a result, "down-regulated". The converse occurs in the case of reduction of pressure in the drive mechanism chamber. As a result, the compressor can be "up-regulated". The corresponding regulating valves in the prior art are electrically controlled. In the process, increasing the pressure within the drive mechanism chamber and, as a consequence, corresponding "down-regulation" of the compressor are accomplished by appropriate opening of a fluid connection between the drive mechanism chamber and the delivery side, or high-pressure side, of the compressor. Arranged in that fluid connection is the mentioned regulating valve, which is preferably electrically controllable. In the process, it should be ensured that the pressure in the drive mechanism chamber does not exceed a predetermined maximum level. For that purpose, a safety fluid connection is provided between the drive mechanism chamber and the suction side of the compressor.

The pressure in the drive mechanism chamber can be adjusted between the high pressure prevailing on the delivery side and the low pressure prevailing on the suction side. The compressor can be up-regulated and down-regulated within those limits. Increasing the pressure in the drive mechanism chamber is of course always accomplished in the prior art in relation to the pressure increase by way of a fluid connection of constant cross-section between the drive mechanism chamber and the suction side of the compressor. In this context it is to be borne in mind that when maintaining the increased differential pressure, owing to the constant cross-section of the mentioned fluid connection, when down-regulating the compressor, that is to say when increasing the differential pressure between the drive mechanism chamber and the suction side, the mass flow flowing out of the drive mechanism chamber becomes continuously and significantly greater. Because that mass flow has to be taken directly from the high-pressure side, it is no longer available in the system for the actual purpose of the compressor, that is to say cooling or heating, and must consequently be regarded as a loss. The mass flow required for down-regulating the compressor is conveyed, almost exclusively by internal compressor means, from the high-pressure side, by way of the regulating valve, to the drive mechanism chamber and from there, through the fluid connection between the drive mechanism chamber and the suction side, back to the suction side, from where is it is again drawn in and compressed. Additional outlay which provides no direct benefit is required for compression of that so-called "regulating mass flow".

By way of example, Fig. 1 illustrates the above-mentioned behaviour. As the pressure difference between the drive mechanism chamber and the suction side increases (X axis), the mass flow through the fluid connection between the drive mechanism chamber and the suction side increases significantly. In addition to the loss mass flow, the associated inlet and outlet pressures before and after the fluid connection or opening between the drive mechanism chamber and the suction side are also shown, as well as, by way of example, a possible temperature plot at the inlet. All the curve plots are to be regarded merely as examples; however, the basic behaviour for all typical operating points of a compressor for a vehicle air-conditioning system can be recognised. The starting point of the mass flow curve, located at a low pressure difference between the drive mechanism chamber and the suction side and at a corresponding low mass flow, is defined substantially by internal leakage and other factors which will not be described in greater detail here. The free cross-section of the fluid connection between the drive

mechanism chamber and the suction side is usually so selected that, for all the operating states that are to be assumed, undesired down-regulation of the compressor does not come about.

Especially in central Europe, with comparatively moderate average annual temperatures and relatively low average atmospheric humidity, air-conditioning systems used in particular in the motor vehicle sector are frequently down-regulated (with the above-mentioned inherent losses caused by the down-regulation). The present invention is intended to provide a simple, efficient and economical solution to that problem in particular.

A further problem besides the energy losses is posed by the loading of the pistons and of the "tilt plate mechanism". The pressure-related main direction of force in compressors is axial, from the upper side of the piston to the underside of the piston. The opposite loading case (in the direction of the upper side of the piston) occurs, as a result of pressure, significantly only in the case of down-regulation of the compressor, that is to say in the case of an increase in pressure in the drive mechanism chamber above the pressure of the suction side. Therefore, in down-regulated operation it must be ensured that the pressure-related forces acting on the underside of the piston do not exceed a defined level. In this context it must also be borne in mind that, preferably, pistons made of light materials, especially light metal, are to be used, which are advantageous both in energy terms and in regulation terms. The present invention is intended to provide a possible solution under this aspect too.

Consequently, the problem underlying the present invention is to minimise the loss mass flow that occurs during down-regulation, in particular in the case of an externally regulated compressor having any regulation characteristic, and, on the other hand, to provide a safety device which is capable of limiting or reducing the pressure-related forces acting in the direction of the upper side of the piston during the intake process.

In the case of a compressor of the kind mentioned at the beginning, the problem is solved in accordance with the invention by means of the fact that there is provided between the drive mechanism chamber and the suction side a fluid connection in which there is arranged a continuously operating regulating valve by means of which, starting from a predetermined pressure difference between the drive mechanism chamber and

the suction side, the fluid connection between the drive mechanism chamber and the suction side is increasingly throttled as the pressure difference further increases and is, in the extreme case, closed completely.

The "constant" opening provided between the drive mechanism chamber and the suction side in the prior art is accordingly replaced, in accordance with the invention, by a "variable opening", more particularly in regulation terms in such a way that the opening cross-section is increasingly reduced as the differential pressure between the drive mechanism chamber and the low-pressure side, or suction side, increases, as a result of which the loss mass flow can be kept almost constant at the original value.

Preferably, the regulating valve in the fluid connection between the drive mechanism chamber and the suction side opens again in the event of a predetermined excessively high pressure difference between the drive mechanism chamber and the suction side so that damage to or destruction of the pistons is countered. Accordingly, the basic concept is that, in the event of an excessively high pressure difference between the drive mechanism chamber and the suction side, at least one opening which is not active in normal operation comes into effect, through which opening a flow of mass out from the drive mechanism chamber is possible in such a way that the pressure in the drive mechanism drops back to a lower operating pressure. This measure is a safety measure in order to protect the compressor, or drive mechanism chamber, from undesirable over-pressure.

Further details of the invention, especially constructional details, are given in claims 3 ff. Reference will also be made to those details separately in the course of the following description of the invention, especially using examples of embodiments. In that respect, reference is made to the accompanying drawings, in which:

Fig. 2 shows the influence of various temperatures on the loss mass flow;

Fig. 3 shows, in diagrammatic form, a first arrangement of a regulating valve constructed and arranged in accordance with the invention, between the drive mechanism chamber and the suction side of a compressor (normal operating state);

Fig. 4 shows the typical down-regulation curve of an externally regulated compressor;

Fig. 5 shows a possible force plot within a mechanical regulating valve according to the invention during down-regulation;

Fig. 6 shows a comparison of the loss mass flows through a fluid connection between the drive mechanism chamber and the suction side, the cross-section of the opening of which is variable by means of the regulating valve according to the invention; and

Fig. 7 shows the arrangement according to Fig. 3 with activated safety function.

If one considers the loss mass flow for typical operating conditions of a compressor, especially an axial piston compressor for vehicle air-conditioning systems, it can be seen that, for a down-regulation range, the influence of various temperatures and inlet conditions is small. Rather, as can be seen from Fig. 2, it is the pressure difference between the inlet side and outlet side of a regulating valve arranged in a fluid connection between the drive mechanism chamber and the suction side that is decisive for the mass flow that flows out.

If a medium condition is taken as dimensioning reference, for the usual down-regulation range generally the maximum difference of the actually occurring "extreme points" is less than about 2 % (in relative terms) for the usual down-regulation range.

The fact that the differential pressure between the drive mechanism chamber and the suction side is substantially responsible for the loss mass flow can be exploited by means of a mechanical regulating valve according to Fig. 3. Accordingly, the regulating valve 10 comprises a cylinder space 11, which has, on the one hand, a fluid connection, by way of a line 12, with the drive mechanism chamber of an axial piston compressor and, on the other hand, a fluid connection, by way of a further line 13, with the suction side of the compressor and within which a piston 14, closed on the low-pressure side, is mounted so as to be displaceable back and forth in each case against the action of a resilient element - in this instance helical compression springs 15, 16 - and of the forces caused by the inlet and outlet pressure, wherein the piston 14, in dependence on the pressure difference acting on the piston 14 corresponding to the pressure difference

between the drive mechanism space (indicated by reference numeral 17 in Fig. 3) and the suction side (indicated by reference numeral 18 in Fig. 3), reduces the effective valve opening between the drive mechanism space and the suction side to a greater or lesser degree, and in the extreme case closes it completely. The mentioned fluid passageway is defined by the lines 12, 13 and also the cylinder space 11 and the piston 14, which for the purpose is in the form of a hollow piston open at one end face (the upper end face in Fig. 3), in the wall 19 of which there is formed at least one axially extending, especially slot-shaped, passageway 20, with which passageway 20 there is associated the suction side 18 or a fluid line 13 in communication with the suction side 18 and opening out laterally into the cylinder space 11. The internal space 21 of the hollow piston 14 has a fluid connection with the drive mechanism chamber 17 by way of the open end face 22. On the suction side, the piston 14 is closed off by a piston base 23. The suction side, that is to say low pressure, is applied externally to that piston base 23. For that purpose, the cylinder space 11 below the piston base 23 is in communication by way of a connecting line 24 with the suction side, or with the line 13 leading to the suction side.

The piston 14 is clamped between two springs, in this instance helical compression springs 15, 16, in contact with its end faces, within the cylinder space 11. The cylinder space 11 is defined by a corresponding bore in a valve body 25, the opening of the bore being closed off by means of a stopper 26 after the helical compression springs and the piston 14 have been put in place.

The spring elements 15, 16 are so constructed and adjusted that the throttling behaviour of the regulating valve 10 with increasing pressure difference between the drive mechanism chamber 17 and the suction side 18 is either linear or progressive, degressive and/or stepped. This is also dependent on the formation of the passageway 20 in the piston 14. The slot-shaped passageway 20 in the wall 19 of the piston 14 can be constructed in the form of a slot that becomes wider or narrower either continuously or stepped in one direction axially, more specifically in accordance with the desired regulation behaviour. In order to achieve a constant mass flow, a geometry that becomes continuously narrower in the axial direction is to be provided.

The valve body 25 can be part of the compressor housing or a separate component. When the piston 14 is made of plastics material, the springs 15, 16 are preferably provided integrally with the piston as a structural unit, that is to say are integrated with

the piston material at the end faces by casting. As already mentioned, the piston 14 is installed within the cylinder space 11 with biasing by the two spring elements 15, 16 so that the springs are in contact with the piston 14 in all operating states.

The piston 14 is fitted inside the cylinder space 11 with a play fit, more specifically preferably with a fit of less than 15 μm, in order to keep the mass flow flowing past the piston at a negligibly low level. In order to achieve this, additional sealing measures can be provided between the piston and the cylinder wall.

In Fig. 3, the piston 14 is constructed with identical end faces. It is also feasible to use a differential piston having end faces of different sizes instead, this being governed by which forces, especially differential forces, are acting on the piston. The spring forces produced by the spring elements 15, 16 are, in contrast, of subordinate importance. In that respect reference is made to Fig. 5.

When the pressure between the drive mechanism chamber and the suction side is balanced, the piston 14 is held in a middle position. The passageway 20 provided in the piston wall 19 is then located at approximately the same height as the line 13 leading to the suction side 18.

Usually a minimum differential pressure between the drive mechanism chamber and the suction side is necessary in order to bring about a reduction in the compressor stroke. This minimum differential pressure should be taken into account when designing the described mechanical regulating valve. Firstly, in terms of design, the slot-shaped passageway(s) 20 incorporated in the piston wall 19 should, as far as possible, be so positioned that, in the case of the mentioned minimum pressure difference, the full area of the passageway(s) 20 is effective. A further increase in the differential pressure should, however, result as immediately as possible in successive reduction of the effective opening cross-section of the slot-shaped passageway(s) 20.

The design of the passageway opening in the piston wall 19 should be such that internal leaks, or other factors which will not be described in greater detail here, can flow out completely by way of the piston 14, or regulating valve 10, in the case of a minimum pressure difference between the drive mechanism chamber and the suction side.

In the course of the down-regulation procedure, the differential pressure between the drive mechanism chamber and the suction side increases. A possible force plot of spring and pressure forces in the regulating valve 10 in the case of increasing differential pressure and corresponding displacement of the piston 14 (downwards in Fig. 3) is shown by way of example in Fig. 5, in which it is also assumed that, in the course of down-regulation, an increase in the low pressure on the suction side also comes about.

The regulating valve 10 can have any desired installation position, because the weight of the piston 14 itself is negligible for regulation. The regulating valve 10 can be arranged at the cylinder head or cylinder block or, taking appropriate connections into account, outside the compressor housing.

As can also be seen from Fig. 5, the regulating piston 14 moves towards the suction side as the differential pressure increases. As a result, the slot-shaped passageway 20 in the piston wall 19 is increasingly covered over and, correspondingly, the opening cross-section is increasingly reduced. The passageway 20 is preferably so constructed that after displacement of the piston 14 there remains a residual opening cross-section such that an almost constant mass flow is established between the drive mechanism chamber and the suction side. In that respect reference is made to Fig. 6.

Because the mass flow flowing out with the aid of the described regulating valve 10 during down-regulation can generally be maintained at an approximately constant low level (see Fig. 6), a significantly lower loss mass flow is required in down-regulated operation in order to establish the differential pressure between the drive mechanism chamber and the suction side that is required for down-regulation. As a result, comparatively less drive power has to be expended for the same cooling or heating performance. The efficiency is consequently increased, more specifically for the entire down-regulation range, the relative loss mass flow being increasingly reduced as the differential pressure between the drive mechanism chamber and the suction side increases.

In the case of the arrangement according to Fig. 7, a further opening, or a so-called safety slot 27, is provided in the piston wall 19 above the slot-shaped passageway 20, that is to say in the direction of the delivery side, which comes into effect immediately when the lower slot-shaped passageway 20 has been completely covered over, more

specifically when the pressure in the drive mechanism space becomes excessively high and the piston 14 in the regulating valve 10 is consequently displaced further to the suction side. The safety slot 27 therefore comes into effect when the differential pressure between the drive mechanism chamber and the suction side reaches a predetermined maximum value. The pressure in the drive mechanism chamber can then be effectively and rapidly reduced to a lower operating pressure by way of the safety slot 27.

In other respects, the regulating valve according to Fig. 7 is constructed correspondingly to that according to Fig. 3, and elements that have already been described with reference to Fig. 3 are indicated in Fig. 7 by the same reference numerals.

As a result of the regulating valve described, a further advantage is also achieved, namely the advantage that the mass flow between the drive mechanism chamber and the suction side of the compressor is greatly reduced. As a result, the oil mass flow, that is to say the amount of oil carried along with the gas flow, is also correspondingly reduced. This on the one hand has a positive effect on the overall performance and also on the thermal behaviour of the compressor and, as a result, that of a vehicle air-conditioning system and on the other hand has an advantageous effect on the service life of the compressor.

The described regulating valve can be provided in the form of a prefabricated structural unit. Further elements, such as oil separators, particle filters or the like, can be integrated within the piston or also within the cylinder space.

The valve body 25 is preferably made of steel, steel alloy, light metal, especially aluminium, or also plastics material. The same applies to the piston 14. When the piston 14 is made of plastics material it is possible for the spring elements 15, 16 to be intimately or permanently connected to the piston material so that the piston and spring elements form a structural unit which can be introduced as one entity into the cylinder space 11. When the piston 14 is made of plastics material, manufacture by injection-moulding, above all, is advantageous. Thermosetting plastics or thermoplastic materials can be used. In the case of injection-moulding, the passageway slots 20 and 27 can be formed in one operation. Plastics material optimised in terms of sliding properties,

especially thermosetting plastics or thermoplastic materials optimised in terms of siliding properties, can be used.

When metallic materials are used for the piston 14, the passageways 20, 27 are formed preferably by laser (laser cutting). By that means any desired contours and opening cross-sections can be obtained. As already mentioned, the slot-shaped passageway 20 in the wall 19 of the piston 14 can become wider or narrower either continuously or stepped in one direction axially, in dependence on the desired regulation behaviour, with preference being given in this case to the narrowing geometry. The contour of the passageway 20 is finally also dependent on the compressor itself and its operating behaviour.

The fluid line 12 in communication with the drive mechanism chamber 17 can of course also be so arranged that it opens out into the cylinder space 11 axially. To that extent, Figs. 3 and 7 are merely diagrammatic representations of the general principle. In other words, the inlet opening into the cylinder space 11 can, if required, also be provided through the closure stopper 26. In the region of the mouth of the line 13, which leads to the suction side, there can be provided an outward bulge, which ensures that gas can flow out through the passageway 20 even when the piston 14 rotates about its longitudinal axis. In corresponding manner, the passageway 20 can also be positioned within an inward bulge, especially an annular inward bulge, in the piston wall 19. It is also possible to provide a plurality of passageways 20 in the piston wall 19, which are arranged distributed over the circumference of the piston. Preference is given, however, to the piston 14 being positioned within the cylinder space 11 so that it is secured against rotation.

The connection between the outside of the piston base 23 and the suction side can, moreover, be made directly by way of the piston 14, more specifically by way of longitudinal grooves formed on the piston or on the cylinder wall. These pressure-communicating grooves are accordingly intended to make a fluid connection between the suction side and the space beneath the piston base 23. In order to ensure that this is the also the case in any desired axial position of the piston 14, the corresponding grooves are preferably formed on the cylinder wall. Pressure-communicating grooves of such a kind also have the advantage of improved axial mobility of the piston 14. Oil carried along by the coolant, which might otherwise impair the mobility of the piston 14

within the cylinder space 11, accumulates within the pressure-communicating longitudinal or axial grooves. By means of the mentioned longitudinal or axial grooves, therefore, the contact area between the piston and the cylinder wall is reduced on the one hand. On the other hand, an oil-collecting space is created so that the oil - especially at low ambient temperatures - does not hinder the mobility of the piston 14 within the cylinder space 11. The risk of the piston 14 being restricted in terms of its freedom to move within the cylinder space 11 is reduced. In addition, it is ensured that the pressure difference between the drive mechanism chamber and the suction side is applied to the piston 14 without having to provide an additional connection with the low-pressure side.

There are three significant positions of the piston 14:

- middle position in the case of pressure balanced between the drive mechanism chamber and the suction side (the opening cross-section of the passageway 20 in the piston wall 19 being fully effective)

- working point with application of pressure difference between the drive mechanism chamber and the suction side (the opening cross-section of the passageway 20 being reduced)

- safety position in the case of an undesirably high pressure difference between the drive mechanism chamber and the suction side (only the safety slot 27 being effective)

All features disclosed in the application documents are claimed as being important to the invention insofar as they are novel on their own or in combination compared with the prior art.

Reference numerals

10 regulating valve
11 cylinder space
12 line
13 line

14	piston
15	helical compression spring
16	helical compression spring
17	drive mechanism space
18	suction side
19	piston wall
20	(slot-shaped) passageway
21	internal space
22	open end face
23	piston base
24	connecting line
25	valve body
26	closure stopper
27	safety slot